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MACHINE TOOL DYNAMIC MEASUREMENTS AND DIAGNOSTIC SYSTEM

R. K. WHARTON

T. P. VINCENT

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Vibrations in machine tools, commonly known as chatter, often are the cause for poor machining operations and breakdown of machinery. A state-of-the-art measurement and analysis system was designed, fabricated, and implemented. The system performs real-time acquisition of raw vibration measurement data and provides analysis for preventative/predictive maintenance action.

(CONT'D ON REVERSE)

7. AUTHORS (CONT'D)

T. P. Vincent
Operations Directorate
Watervliet Arsenal
Watervliet, NY 12189-4050

20. ABSTRACT (CONT'D)

The system has been used to determine the causes for poor surface finishes occurring during metal removal processes. In the first case, during rough and finish grinding operations, several grinder spindles were found to be significantly out of balance. Investigation revealed that during balancing of spindles, the spindle manufacturer was using frequency versus mils engineering units to represent when a spindle was balanced. This procedure was found to be inadequate for the grinder's surface finish production requirement. The system's real-time capability resulted in the spindle manufacturer's redesign of his spindles and the acceptance of frequency versus g' level vibration units as more precise parameters to represent when a spindle balanced condition was present. In the second case, a surface grinder was producing poor surface finishes due to excessive vibration chatter attributed to spindle ball bearing outer race defects.

The implementation of the Machine Tool Dynamic Measurements and Diagnostic System into maintenance practices enables technological advantages in the following manufacturing areas: ability to quickly determine and remedy existing machine tool mechanical problems, reference-baseline vibration signature (newly acquired machine tools and existing units), assistance in maintaining ordnance dimensional and surface finish requirements, improvement in reducing machine downtime, and input for short/long-term management purposes. Keywords

TABLE OF CONTENTS

TABLES

LIST OF ILLUSTRATIONS

TABLE OF CONTENTS (cont'd)

INTRODUCTION

When operating, all machines generate characteristic mechanical vibrations which inherently contain information that can provide an insight to operating dynamics and adequacy of design. The obscured information in time-varying signals can usually be uncovered through vibration spectrum analysis involving the study of dynamic signal amplitudes over a range of frequencies in which the machinery-induced vibrations take place. These major vibration components which are resident in a composite frequency spectrum, often can be traced to specific excitation sources by relating corresponding frequencies to typical design parameters of machine parts.

STATEMENT OF THE PROBLEM

External surface and internal contour grinders are examples of high production machine tools in use at the Watervliet Arsenal.

During metal removal processes, vibrations in machine tools frequently are the cause of poor workpiece surface finishes. This condition, if left unchecked, generates lost time and dollars, particularly in a high production environment. It is vital to bring about timely awareness and determination of the origin and causes that produce these vibrations.

OBJECT

A technique for measuring and establishing machine tool baseline vibration signatures was needed. An analysis of the vibration spectra would yield causal and location of defective machine tool components.

Subsequent analysis would indicate whether a machine tool required corrective action, i.e. whether the machine spindle should be balanced or the thrust bearing checked.

A diagnostic system was developed and designed for measuring the raw analog vibration signals emanating from machine tools during their use. The system is able to categorize and identify suspect components and suggest steps that should be taken to place a machine tool into optimum performance. These actions have been derived from vibration criteria and standards adopted by the American National Standards Institute (ANSI) and the Vibration Institute.

PROCEDURE

Machine Tool Dynamic Measurements and Diagnostic System

Design

The Machine Tool Dynamic Measurements and Diagnostic System consists of four separate subsystems: sensors (accelerometers), FM four-channel tape recorder, real-time dynamic signal analyzer, and computer hardware/ software. The total system monitors, detects, measures, and analyzes vibration spectra. Figure 1 is representative of the total system concept.

Operational sequence

Initial "baseline" vibration measurements are obtained at the machine tool site under no-load, run-up, and run-down operating conditions. Accelerometers are mounted externally on the machine tool housing and are primarily oriented in the horizontal, vertical, and axial

directions. When a questionable vibration reading or placement position for an accelerometer is encountered, a radial location is selected. During operation of the machine tool, voltages are generated within the accelerometer by changes in forces acting upon the accelerometer's piezoelectric crystal. These signals are proportional to acceleration (force = mass times acceleration).

Signal measurement

Two methods were employed to obtain "raw" analog vibration signals.

Method 1: The spectral voltages are simultaneously recorded on three of the tape recorder's four channels. The remaining channel is used to narrate conditions occurring during the measurement sequence, i.e. accelerometer calibration voltages and machine tool operating speed. One at a time, each channel of taped analog data is entered into the analyzer which has been pre-programmed to accept the signals and perform digital conversion and analysis. Signal conditioning functions, i.e. low and high frequency band bypass filters, are also built into the analyzer, and amplitude accuracy/frequency resolution techniques organize the converted digital signals into appropriate form. Fast Fourier transform equations perform mathematical operations on the data and derive the equivalent analog waveforms. When requested, the processed data records can be displayed on the analyzer's CRT screen.

Method 2: During the course of the program, the procedure for acquiring and converting raw measurement data was improved by reducing the number of required steps. This was accomplished by direct signal input from the accelerometers to the digital signal analyzer.

Data Reduction and Analysis

Subsequent to using either Method 1 or 2, a special purpose software program "Alert," resident to the computer, retrieves the machine tool's latest processed spectral data from the analyzer's bubble memory and compares these data with prior spectral records stored on floppy disc. Numerical tables of vibration frequencies and amplitudes are issued and a current trended "signature" is derived. The updated data tables and "signature" are copied on the mini-disc to form a permanent archive.

All future measurements are compared with the initial baseline frequency signature and amplitude (i.e. g'level) thresholds to determine if significant changes have occurred. When values have exceeded the established boundary conditions, or new suspect frequencies have surfaced, management reports are generated specifying appropriate corrective action.

DISCUSSION OF RESULTS

The overall g'levels for computed frequencies were obtained from criteria specified in a published article (Reference 1). Rolling element defect equations and their calculations are included in Appendix A, and were used to determine the probable spindle ball bearing assembly vibration defect frequencies.

1. S. M. Norris, "Suggested Guidelines for Forced Vibration in Machine Tools for Use in Protective Maintenance and Analysis Applications," Vibration Analysis to Improve Reliability and Reduce Failure, ASME, New York, NY, 1985, pp. 77-82.

A discussion follows describing two applications for which the system was employed to resolve specific mechanical/surface finish problems.

In the first case, Internal Powder Chamber - 120-mm ordnance system, during rough and finish grinding operations, several grinders were producing an undesirable 64-microinch surface finish. Each grinder spindle (Figure 2) was instrumented with accelerometers and vibration testing was performed.

Analysis of the vibration spectra suggested that all of the spindles monitored, though previously balanced by the spindle manufacturer, were still out of balance. It was determined that the spindle Original Equipment Manufacturer (OEM) and other contractors commissioned to provide these services were utilizing mediocre balancing criteria, i.e. mils. This parameter was not accurate enough for the grinder to produce the surface finish accuracy requirements mandated for this machining operation.

A closer look at one machine tool's spindle vibration measurements also revealed the presence of additional mechanical faults. The defect calculations in Appendix A for ball bearing assemblies, Figure 3, suggest that two bearing assemblies on the same spindle register defects at 815 Hz and 805 Hz. These frequencies are associated with the outer race of bearing at spindle location No. 1, and inner race of bearing at spindle location No. 4, respectively.

Through further testing and analysis, and using a test model of (0.1 to 0.2 g') threshold levels, the spindle was balanced and the required

16-microinch surface finish was obtained. Analog waveform plots for these defect frequencies are shown in Figures 4 through 6. The capability afforded by the digital signal analyzer influenced the spindle manufacturer to redesign his spindles and employ real-time analysis criteria for future balancing of spindles. The remainder of the spindles were tested and balanced as the example stated.

In the second case, during the fabrication of repair parts and gauges (105-mm, 120-mm, 155-mm, and 8-inch ordnance systems) a surface grinder was producing an undesirable 32-microinch surface finish. The (with contact) defect equations in Appendix A and real-time digital signal analysis methodology again were employed. A computed frequency of 122.68 Hz suggested that due to excessive vibration chatter, component wear had occurred in the spindle's ball bearing outer race assembly. The analog waveform plot of this defect is shown in Figure 7. On this occasion, the magnitude of g'levels necessitated removal of the spindle from the grinder for examination and replacement of the ball bearing assembly. Following reassembly of the spindle in the grinder, rebalancing was performed and the required 16-microinch surface finish was attained.

For the applications described, it should be noted that the 0.1 to 0.2 g'level boundary conditions were experimentally determined and have been adopted as appropriate engineering units to represent spindle balance and bearing threshold limits.

CONCLUSIONS

The Machine Tool Dynamic Measurements and Diagnostic System has been demonstrated satisfactorily on internal contour and external surface grinders and incorporated into Maintenance Program Management at Watervliet Arsenal.

The application of this system results in the following benefits:

1. Machine tool diagnostics and timely parts acquisition.

2. Reduction of machine downtime.

3. Input for short/long-term planning (maintenance management purposes).

4. Baseline vibration signature - new machine tool purchases.

5. Improved ordnance dimensional and surface finish requirements.

An experienced user can effectively employ the system for diagnosing and pinpointing problems in both major and subassembly machine tool components. This capability greatly lessens the amount of machine tool disassembly and associated downtime (Table I).

The system is the forerunner of the pilot on-line MACHINERY CONDITION SURVEILLANCE SYSTEM (Figures 8 and 9), and will function as a backup for the plant-wide application.

The Machine Tool Dynamic Measurements and Diagnostic System provides a means for monitoring critical production machinery not included under the on-line plant-wide monitoring system.

It is recommended that vibration measurement and testing criteria be included as part of the acceptance requirements for new machine tool purchases. Also, it is suggested that testing be performed both prior

to shipment of a unit at the contractor's facility and periodically on-site during the warranty period. Vibration "signatures" exhibiting excessive g'levels during on-site field testing could be the basis for acceptance/rejection of newly acquired units.

REFERENCES

1. S. M. Norris, "Suggested Guidelines for Forced Vibration in Machine Tools for Use in Protective Maintenance and Analysis Applications," Vibration Analysis to Improve Reliability and Reduce Failure, ASME, New York, NY, 1985, pp. 77-82.

TABLE I

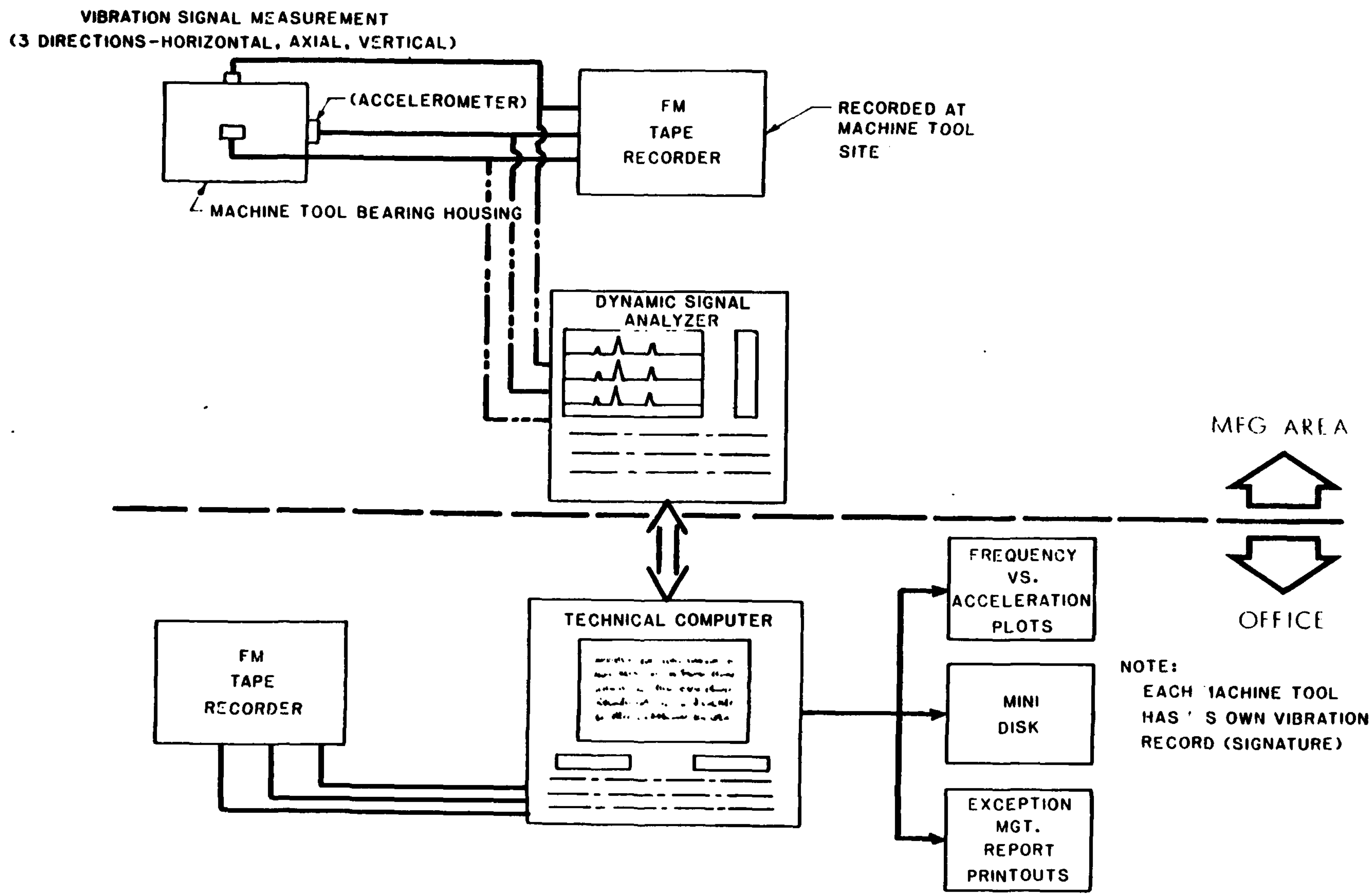
VIBRATION FREQUENCIES AND THEIR LIKELY CAUSES

Frequency in Terms of RPM	Most Likely Causes	Other Possible Causes and Remarks
1/2 x Running RPM	Oil whirl (whip)	1. Bad drive belts 2. Background vibration 3. Subharmonic resonance 4. "Beat" vibration
1-2 x Running RPM	Unbalance	1. Eccentric journals, gears or pulleys 2. Misalignment or bent shaft - if high axial vibration 3. Bad belts if RPM of belt 4. Resonance 5. Reciprocating forces 6. Electrical problems 7. Rubbing shaft
2 x Running RPM	Mechanical looseness	1. Misalignment in high axial vibration 2. Reciprocating forces 3. Resonance 4. Bad belts if 2 x RPM of belt 5. Looseness
2-3 x Running RPM	Misalignment	Usually a combination of misalignment and excessive axial clearances (looseness)
4-8 x Running RPM	Defective roller bearing	
20-50 x Running RPM	Anti-friction bearing	Non-harmonically related

TABLE I (cont'd)

VIBRATION FREQUENCIES AND THEIR LIKELY CAUSES

Frequency in Terms of RPM	Most Likely Causes	Other Possible Causes and Remarks
No. of Vanes x Running RPM	Impeller vanes	Harmonically related
No. of Teeth x Running RPM	Gear Noise	Harmonically related
No. of Blades x Running RPM	Blade rub (N blade)	Harmonically related
Random RPM	Pump cavitation	
Unsteady	Poor foundation	
Vibration soars at one particular speed	Resonance	



CONFIGURATION
FIGURE 1

MACHINE TOOL DYNAMIC MEASUREMENTS AND DIAGNOSTIC SYSTEM

Nomenclature - <u>MACHINE TOOL DYNAMIC MEASUREMENTS AND DIAGNOSTIC SYSTEM</u>

<u>Hewlett Packard Company</u>

Real-time Analyzer - Model 3561A - Single Channel - Range 125μHz - 100 KHz

Desktop Computer - Model 9836C - 1.2 MEGA Bytes

Dual Disc Drives - 3-1/2" - Model 9122D

Dual Disc Drives - 5-1/4" - Model 9134A

Plotter - 2 Pen - Model 7470

Printer - Model 225A

Software - Basic 2.0 and 4.0 Operating Systems

Tape Recorder - FM-4 Channel - <u>Bruel & Kjaer</u> - Model 7006

Accelerometers - ICP Quartz - <u>PCB Co</u>.

Range 2 HZ - 10,000 Hz
" 0.025 Hz - 800 Hz (Seismic)
" 5 Hz - 10,000 Hz

Vibration Envelope Detector - <u>Shaker Research/MTI</u> - Model 223A

Software "Alert" Predictive Maintenance Monitoring System - <u>Structural Measurement Systems</u>

Figure 1A

(ACCELEROMETER) AND BEARING LOCATIONS
SPINDLE

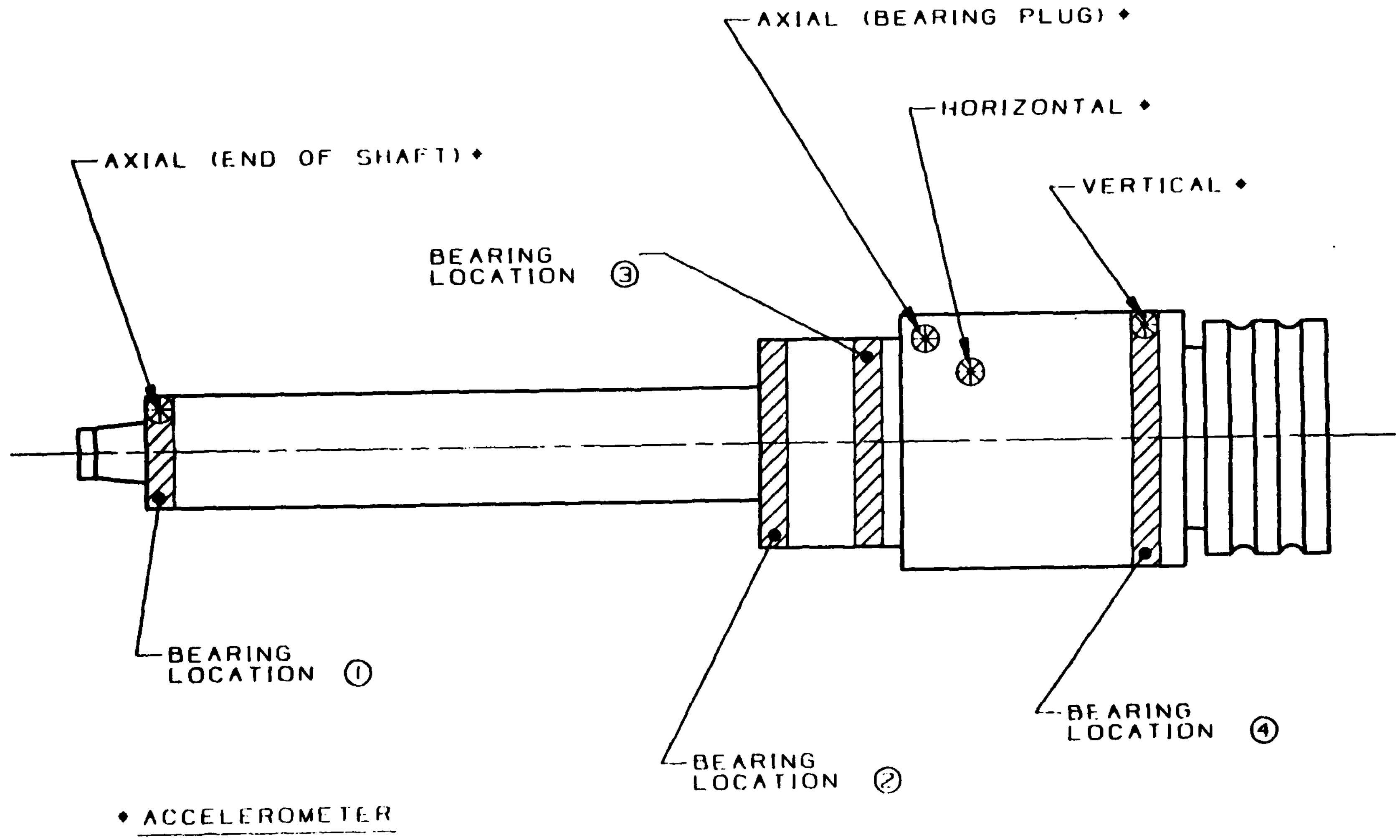
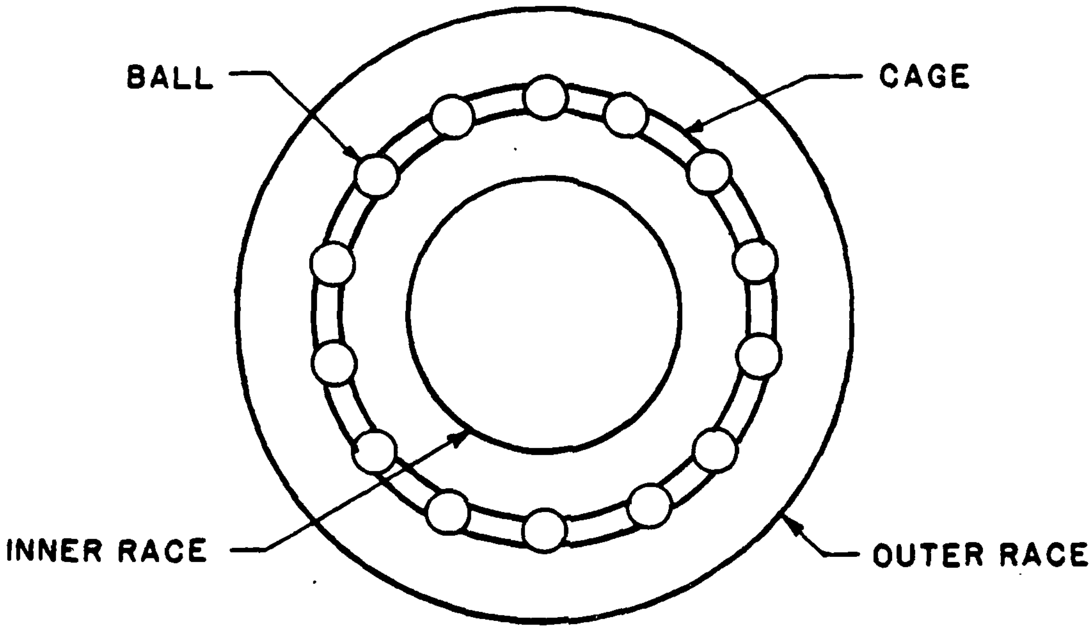


FIGURE 2



BALL BEARING ASSEMBLY

FIGURE 3

GRINDER— **INTERNAL CONTOUR**

ACCELEROMETER LOCATION-HORIZONTAL (Analog Vibration Spectra)

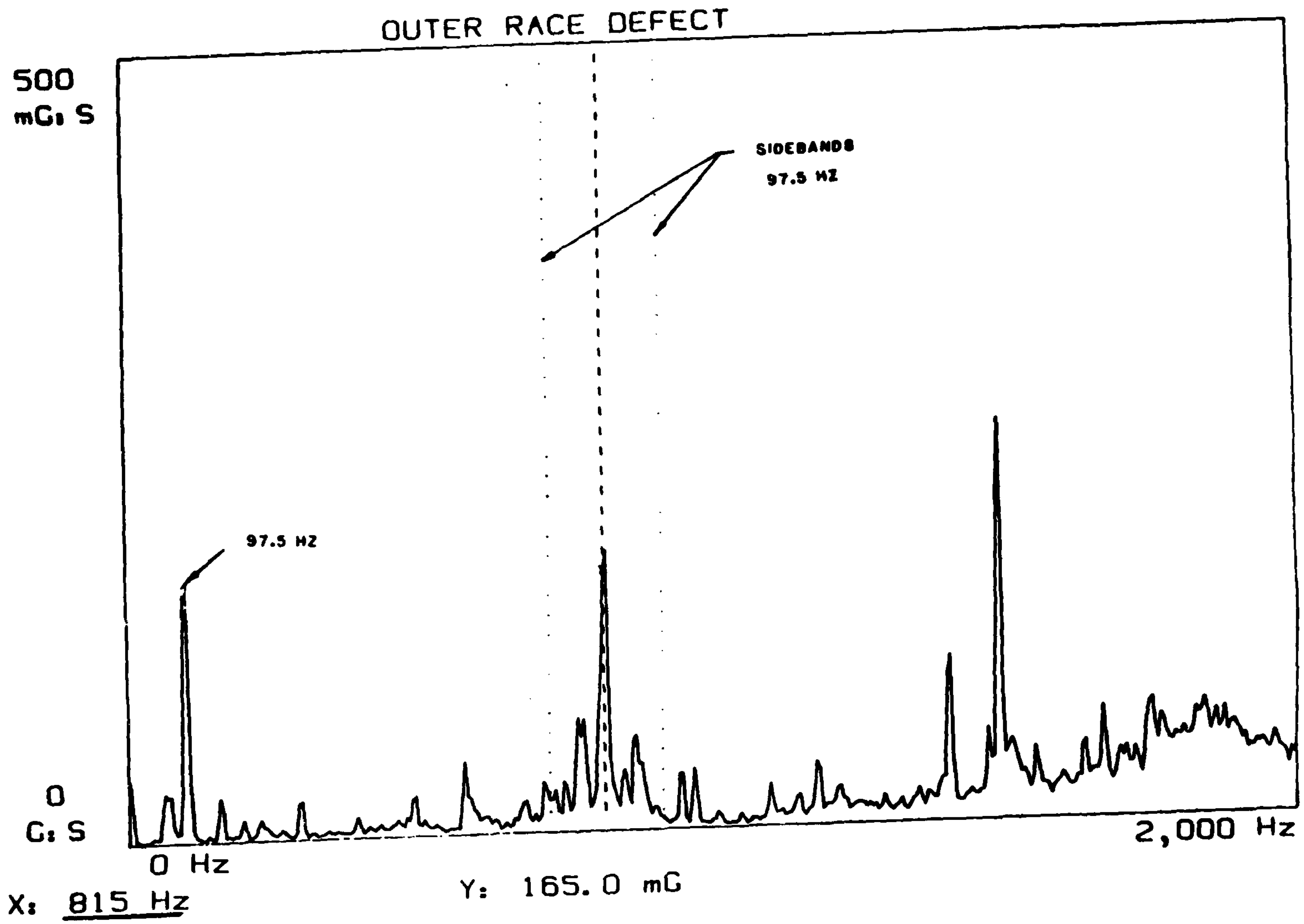


FIGURE 4

GRINDER—INTERNAL CONTOUR

ACCELEROMETER LOCATION-VERTICAL (Analog Vibration Spectra)

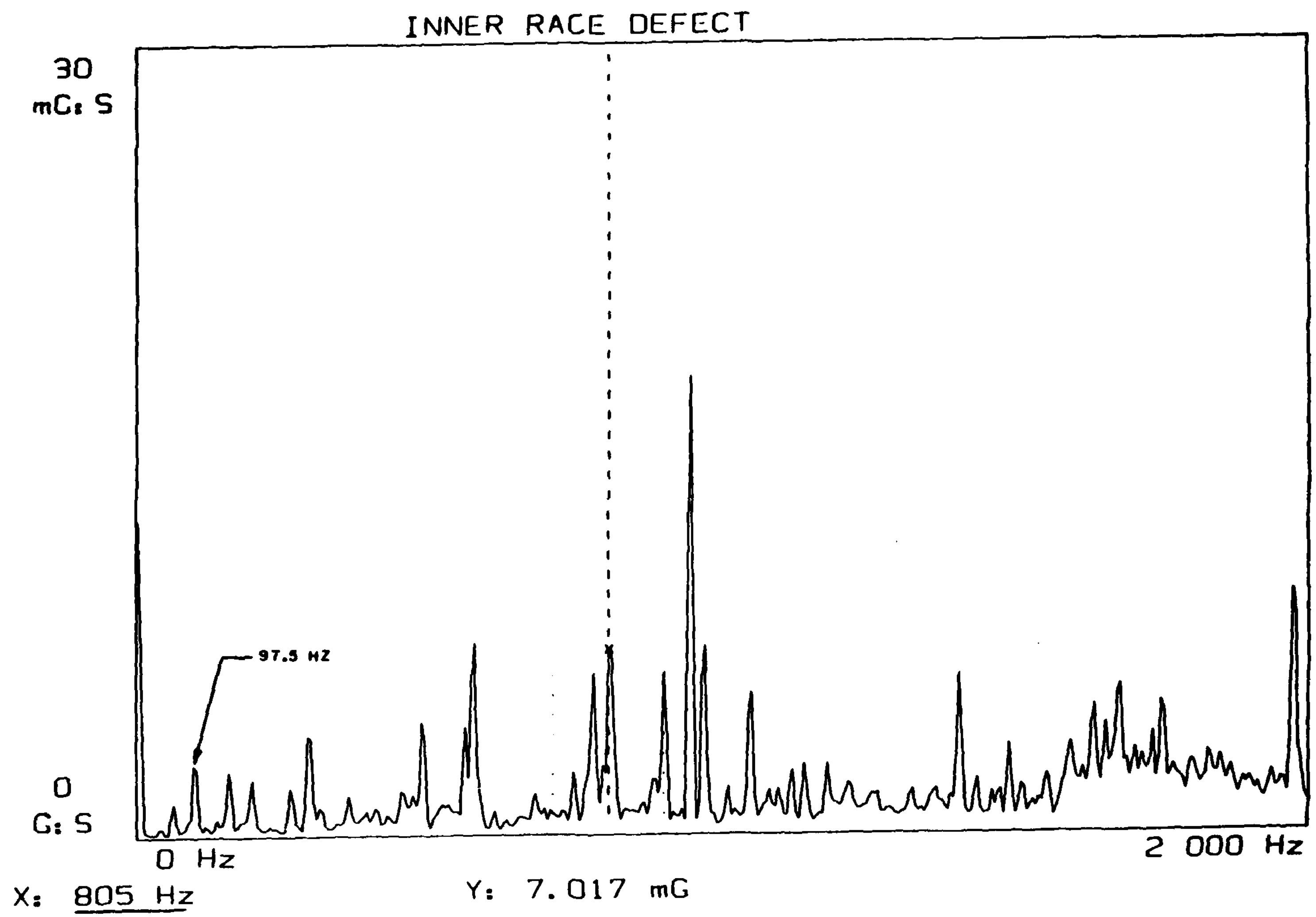


FIGURE 5

GRINDER—INTERNAL CONTOUR

ACCELEROMETER LOCATION—AXIAL (Analog Vibration Spectra)

INNER RACE DEFECT

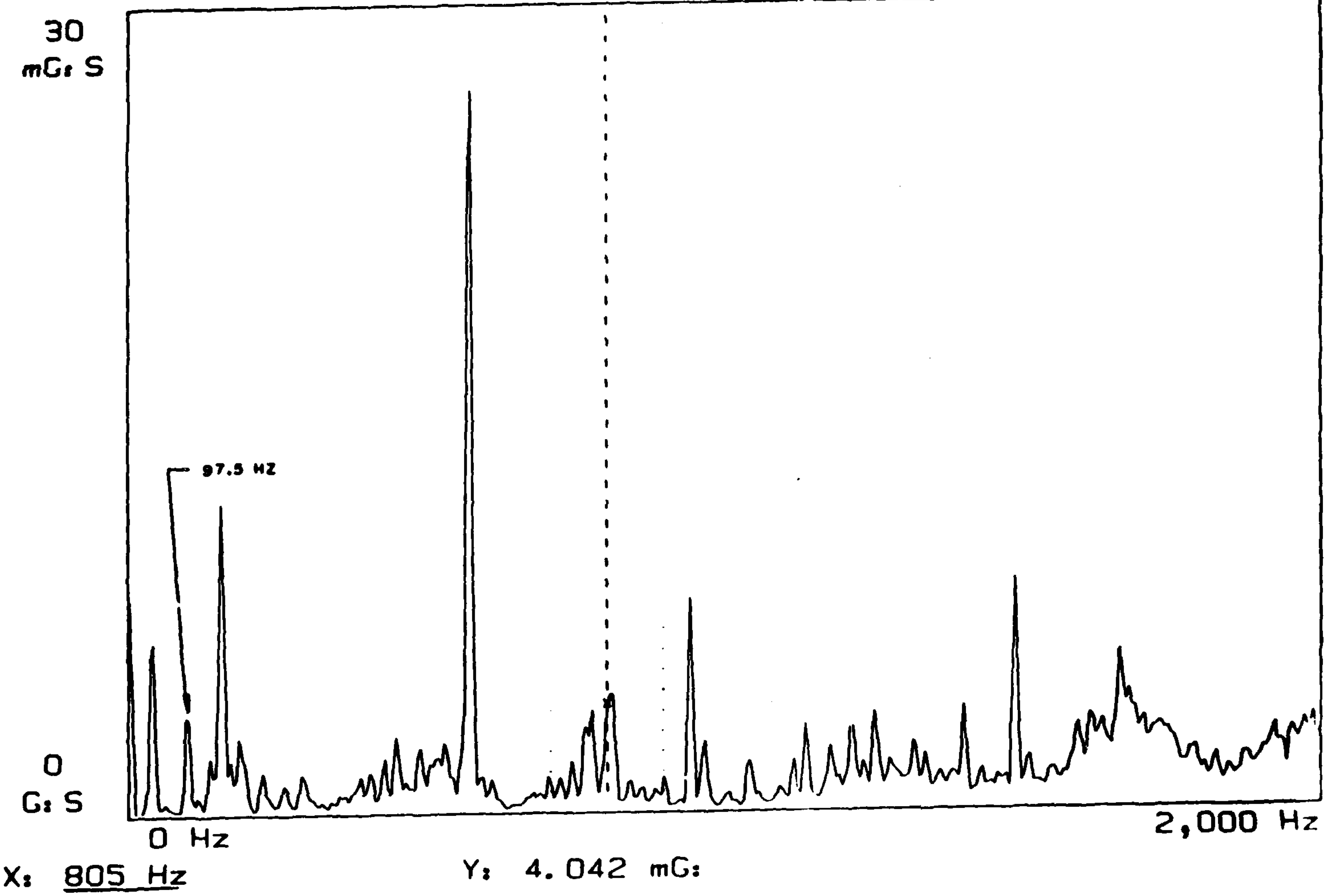


FIGURE 6

EXTERNAL SURFACE GRINDER
ACCELEROMETER LOCATION–VERTICAL (Analog Vibration Spectra)

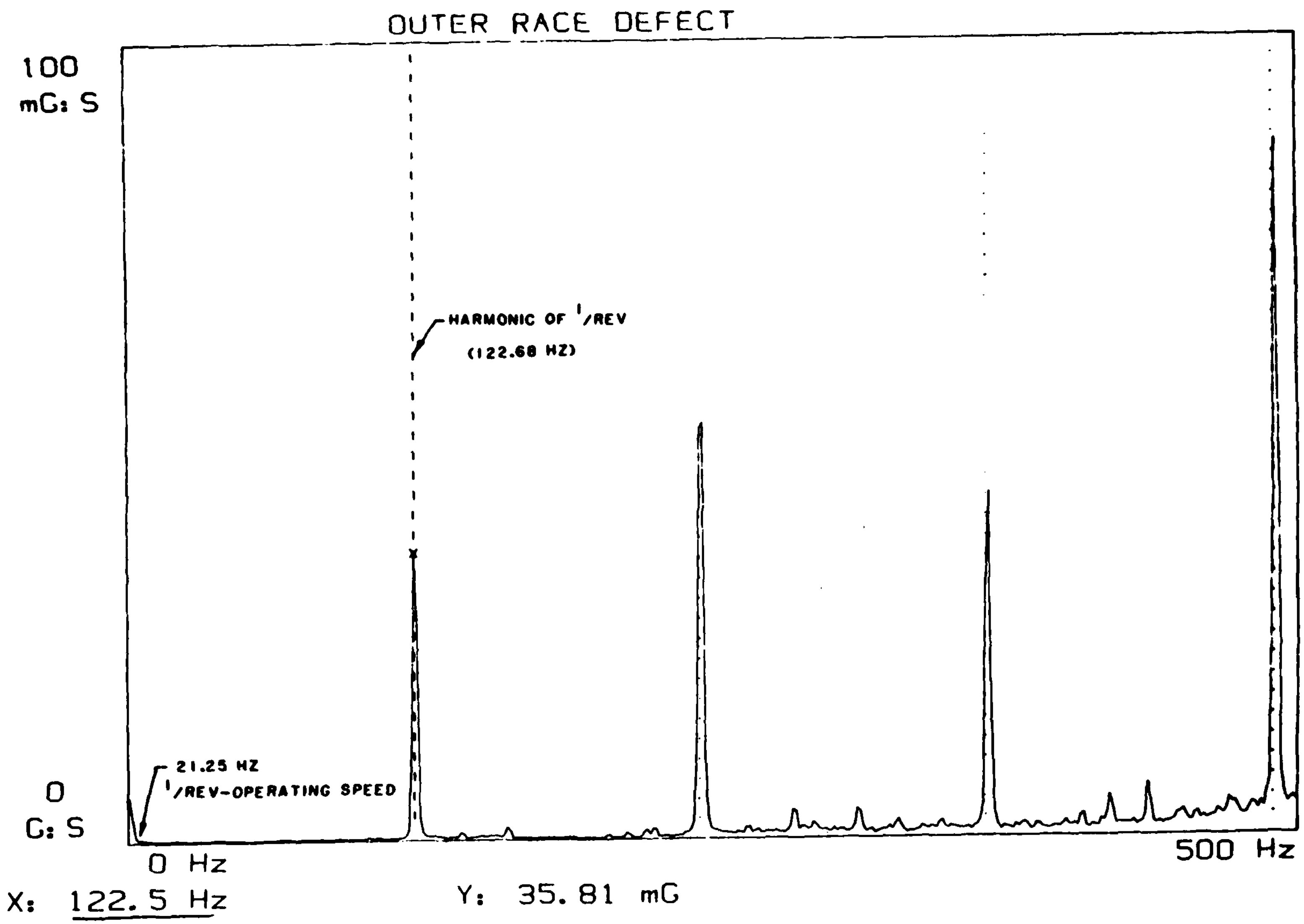


FIGURE 7

MACHINERY CONDITION SURVEILLANCE SYSTEM (MCSS)
PROPOSED PILOT SYSTEM CONFIGURATION
DNC/MCSS LINK

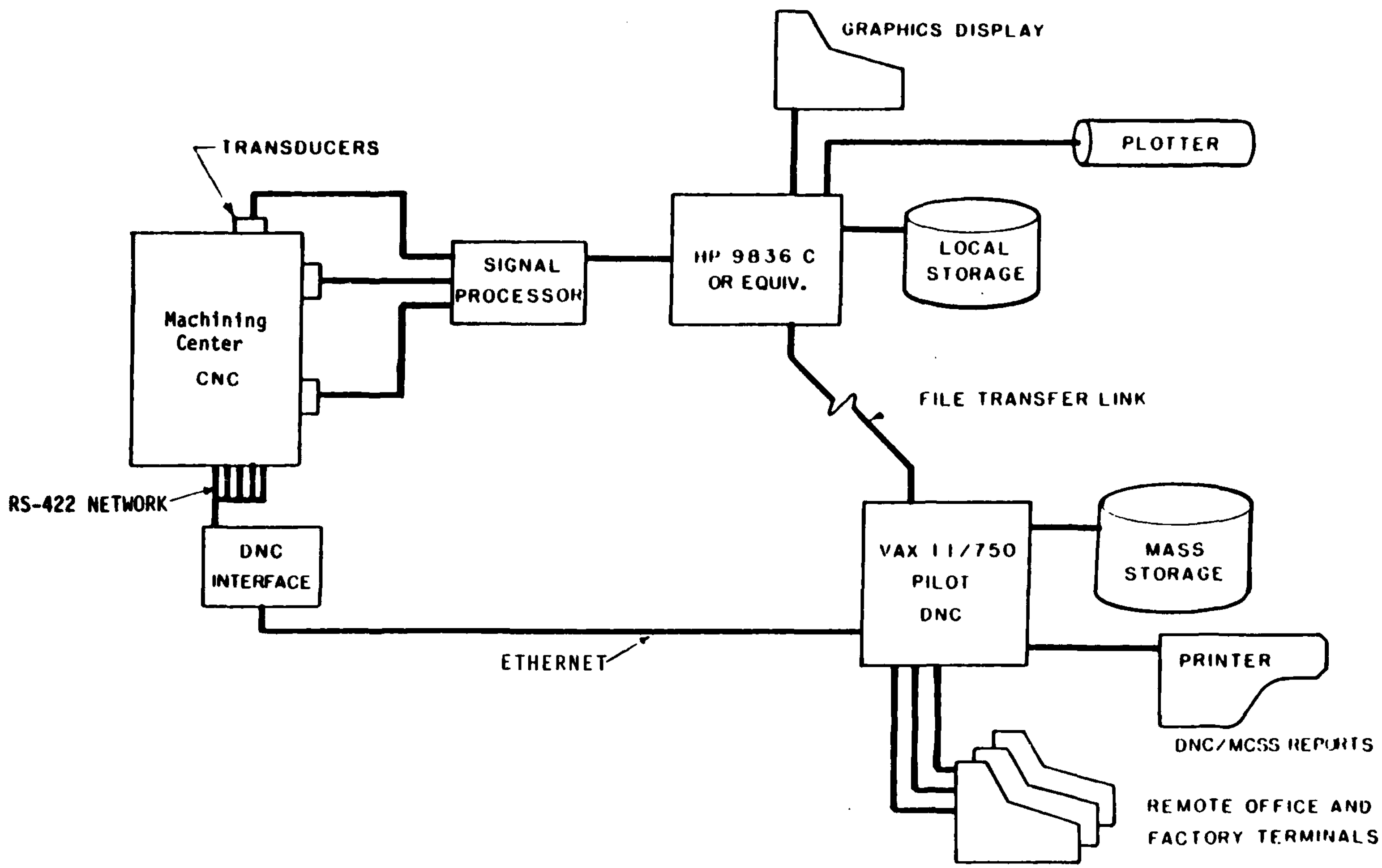


FIGURE 8

MACHINERY CONDITION SURVEILLANCE SYSTEM (MCSS)
PROPOSED FULL SCALE SYSTEM CONFIGURATION
DNC/MCSS INTEGRATION*

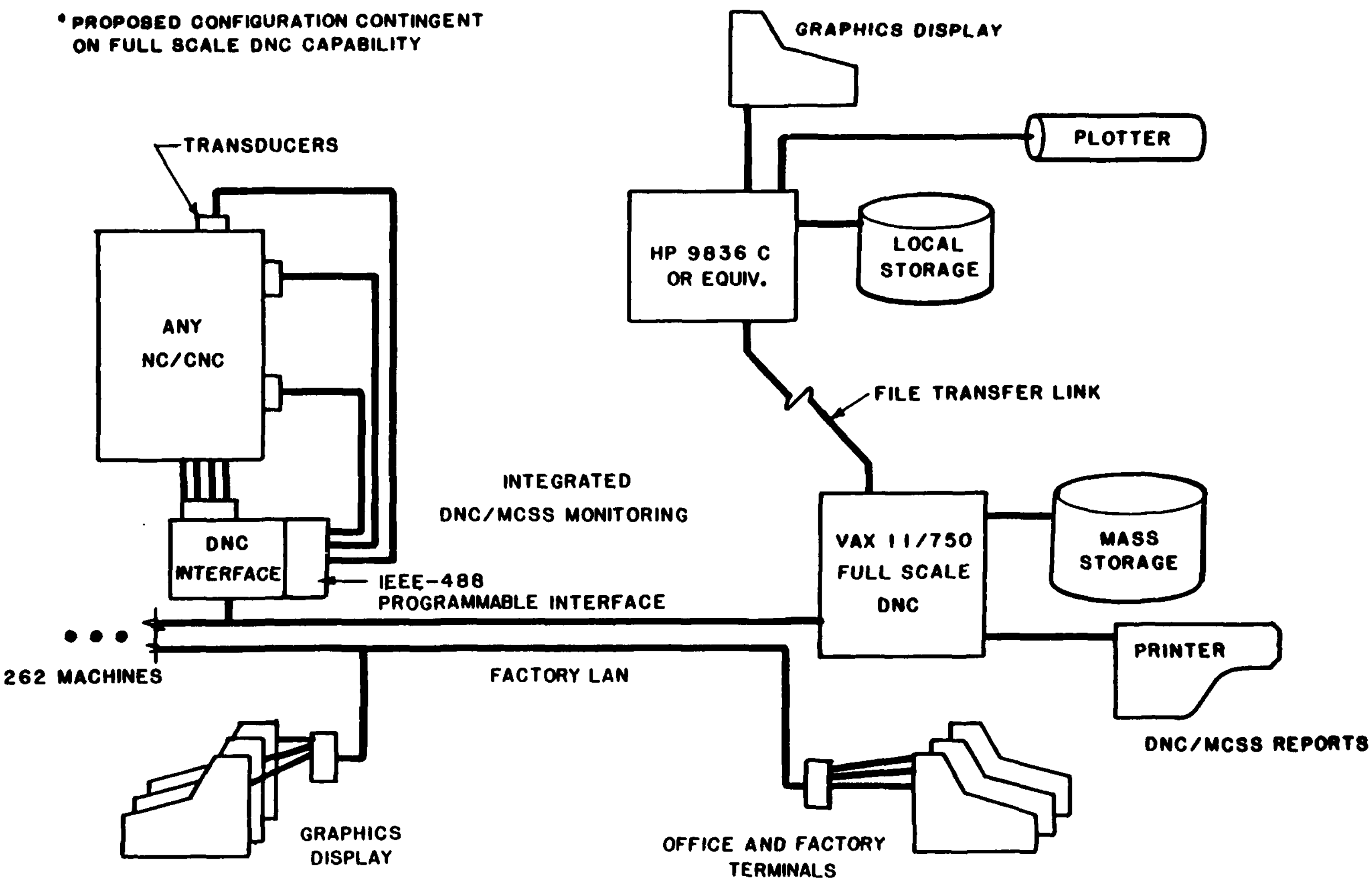


FIGURE 9

DEFECT EQUATIONS

ROLLING ELEMENT BEARINGS

N = Number of Balls

BD = Diameter of Ball

PD = Diameter of Ball Pitch

f_o = Frequency Defect at Outer Race

f_s = Frequency Defect (Spin of Ball)

d_1 = Diameter of Inner Race

d_2 = Diameter of Outer Race

f_r = Frequency (Hz) = RPM/60

f_i = Frequency Defect - Inner Race

α = Ball Angle of Contact with Inner Race

WITH CONTACT

$$f_o = \frac{N}{2} f_r \left(1 - \frac{BD}{PD} \cos \alpha\right)$$

$$f_i = \frac{N}{2} f_r \left(1 + \frac{BD}{PD} \cos \alpha\right)$$

$$f_s = 1/2\, f_r \left(\frac{PD}{BD}\right)\left[1 - \left(\frac{BD}{PD}\right)^2 \cos^2 \alpha\right]$$

WITHOUT CONTACT

$$f_o = Nf_r \left(\frac{d_1}{d_1+d_2}\right)$$

$$f_i = Nf_r \left(\frac{d_2}{d_1+d_2}\right)$$

$$f_s = \left(\frac{d_2}{BD}\right)\left(\frac{d_2}{d_1+d_2}\right)$$

CAGE FAULT

N_1 = No. of Revs/Cage Rev.

$$f_o = Nf_r \left(\frac{1}{N_1}\right)$$

$$f_i = Nf_r \left(1 - \frac{1}{N_1}\right)$$

$$f_s = \frac{f_r}{N_1} \left(\frac{d_2}{BD}\right)$$

$$f_s = 1/2 \left(\frac{RPM}{60}\right)\left(1 - \frac{BD}{PD} \cos \alpha\right)$$

APPENDIX A (Continued)

(INTERNAL CONTOUR GRINDER) ROLLING ELEMENT BEARING DEFECT CALCULATIONS

OUTER-RACE DEFECT (f_o)	BEARING LOCATION
$f_o = \left(\frac{N}{2}\right) \left(f_r\right) \left(1 - \frac{BD}{PD} \text{Cos } \alpha\right)$	
$= \left(\frac{19}{2}\right) \left(97.5\right) \left(1 - \frac{.4680}{3.5433} \text{Cos } 15°\right) = 808 \text{ Hz}$	3
$= \left(\frac{19}{2}\right) \left(97.5\right) \left(1 - \frac{.2188}{1.726} \text{Cos } 15°\right) = \underline{816 \text{ Hz}}$	$\underline{1*}$
$= \left(\frac{14}{2}\right) \left(97.5\right) \left(1 - \frac{.6250}{3.34645} \text{Cos } 15°\right) = 560 \text{ Hz}$	4
$= \left(\frac{14}{2}\right) \left(97.5\right) \left(1 - \frac{.2812}{1.67325} \text{Cos } 15°\right) = 572 \text{ Hz}$	2

INNER-RACE DEFECT (f_i)	
$f_i = \left(\frac{N}{2}\right) \left(f_r\right) \left(1 + \frac{BD}{PD} \text{Cos } \alpha\right)$	
$= \left(\frac{19}{2}\right) \left(97.5\right) \left(1 + \frac{.4680}{3.5433} \text{Cos } 15°\right) = 1044 \text{ Hz}$	3
$= \left(\frac{19}{2}\right) \left(97.5\right) \left(1 + \frac{.2188}{1.776} \text{Cos } 15°\right) = 1036 \text{ Hz}$	1
$= \left(\frac{14}{2}\right) \left(97.5\right) \left(1 + \frac{.6250}{3.34645} \text{Cos } 15°\right) = \underline{805 \text{ Hz}}$	$\underline{4*}$
$= \left(\frac{14}{2}\right) \left(97.5\right) \left(1 + \frac{.2812}{1.67325} \text{Cos } 15°\right) = 793 \text{ Hz}$	3

* SEE FIGURES 4, 5, and 6

(INTERNAL CONTOUR GRINDER)
ROLLING ELEMENT BEARING
DEFECT CALCULATIONS

BALL - SPIN DEFECT (f_s)

$$f_s = \frac{1}{2}\left(f_r\right)\left(\frac{PD}{BD}\right)\left[1 - \left(\frac{BD}{PD}\right)^2 \operatorname{Cos}^2 \alpha\right]$$

	BEARING LOCATION
$= \frac{1}{2}\left(97.5\right)\left(\frac{3.5433}{.4680}\right)\left[1 - \left(\frac{.4680}{3.5433}\right)^2 \left(\operatorname{Cos} 15°\right)^2\right] = 363 \text{ Hz}$	3
$= \frac{1}{2}\left(97.5\right)\left(\frac{1.7760}{.2188}\right)\left[1 - \left(\frac{.2188}{1.776}\right)^2 \left(\operatorname{Cos} 15°\right)^2\right] = 390 \text{ Hz}$	1
$= \frac{1}{2}\left(97.5\right)\left(\frac{3.34645}{.6250}\right)\left[1 - \left(\frac{.6250}{3.34645}\right)^2 \left(\operatorname{Cos} 15°\right)^2\right] = 253 \text{ Hz}$	4
$= \frac{1}{2}\left(97.5\right)\left(\frac{1.67325}{.2812}\right)\left[1 - \left(\frac{.2812}{1.67325}\right)^2 \left(\operatorname{Cos} 15°\right)^2\right] = 282 \text{ Hz}$	2

CAGE SPEED - FUNDAMENTAL TRAIN DEFECT (f_t)

$$f_t = \frac{1}{2}\left(\frac{RPM}{60}\right)\left(1 - \frac{BD}{PD} \operatorname{Cos} \alpha\right)$$

	BEARING LOCATION
$= \frac{1}{2}\left(\frac{5850}{60}\right)\left(1 - \frac{.4680}{3.5433} \operatorname{Cos} 15°\right) = 43 \text{ Hz}$	3
$= \frac{1}{2}\left(\frac{5850}{60}\right)\left(1 - \frac{.2188}{1.776} \operatorname{Cos} 15°\right) = 43 \text{ Hz}$	1
$= \frac{1}{2}\left(\frac{5850}{60}\right)\left(1 - \frac{.6250}{3.34645} \operatorname{Cos} 15°\right) = 40 \text{ Hz}$	4
$= \frac{1}{2}\left(\frac{5850}{60}\right)\left(1 - \frac{.2812}{1.67325} \operatorname{Cos} 15°\right) = 41 \text{ Hz}$	2

APPENDIX A (Continued)

EXTERNAL SURFACE

GRINDER BEARING SPECIFICATIONS

I.D.	- 3.937 Inches
O.D.	- 7.0866 Inches
Number of Balls	- 14
Ball Diameter	- 1.00 Inch
Pitch Diameter	- $\frac{\text{I.D.} + \text{O.D.}}{2}$ - 5.5118 Inches
Contact Angle	- 15°

Grinder Operating Speed - 1/Rev. - 21.25Hz (1275 RPM)

CALCULATED DEFECT FREQUENCIES

<u>Outer Race Defect</u>	<u>- 122.68</u>	<u>Hz</u> *
Inner Race Defect	- 174.8	Hz
Ball Spin Defect	- 56.8	Hz
Cage Defect	- 8.8	Hz

* See Figure 7

EXTERNAL SURFACE GRINDER
ROLLING ELEMENT BEARING
DEFECT CALCULATIONS

OUTER RACE DEFECT (f_o)

$$f_o = \left(\frac{N}{2}\right)\left(f_r\right)\left(1 - \frac{BD}{PD}\ \mathrm{Cos}\ \alpha\right)$$

$$= \left(\frac{14}{2}\right)\left(21.25\right)\left(1 - \frac{1.00}{5.5118}\ \mathrm{Cos}\ 15°\right) = \underline{122.68\ \mathrm{Hz}^*}$$

INNER RACE DEFECT (f_i)

$$f_i = \left(\frac{N}{2}\right)\left(f_r\right)\left(1 + \frac{BD}{PD}\ \mathrm{Cos}\ \alpha\right)$$

$$= \left(\frac{14}{2}\right)\left(21.25\right)\left(1 + \frac{1.00}{5.5118}\ \mathrm{Cos}\ 15°\right) = 174.8\ \mathrm{Hz}$$

BALL - SPIN DEFECT (f_s)

$$f_s = \frac{1}{2}\left(f_i\right)\left(\frac{PD}{BD}\right)\left[1 - \left(\frac{BD}{PD}\right)^2 \mathrm{Cos}^2\ \alpha\right]$$

$$= \frac{1}{2}\left(21.25\right)\left(\frac{5.5118}{1.00}\right)\left[1 - \left(\frac{1.00}{5.5118}\right)^2\left(\mathrm{Cos}\ 15°\right)^2\right] = 56.76\ \mathrm{Hz}$$

CAGE SPEED - FUNDAMENTAL TRAIN (F_t)

$$F_t = \frac{1}{2}\left(\frac{RPM}{60}\right)\left(1 - \frac{BD}{PD}\ \mathrm{Cos}\ \alpha\right)$$

$$= \frac{1}{2}\left(\frac{1275}{60}\right)\left(1 - \frac{1.00}{5.5118}\ \mathrm{Cos}\ 15°\right) = 8.76\ \mathrm{Hz}$$

* SEE FIGURE 7

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